METHOD AND APPARATUS FOR ACHIEVING TEMPERATURE UNIFORMITY AND HOT SPOT COOLING IN A HEAT PRODUCING DEVICE

Related Applications

This patent application claims priority under 35 U.S.C. 119(e) of the co-pending U.S. Provisional Patent Application, Serial No. 60/462,245, filed April 11, 2003 and entitled "**RING STIFFENER PROTECTOR AND REMOVEABLE SPREADER LID**", which is hereby incorporated by reference. This patent application also claims priority under 35 U.S.C. 119(e) of the co-pending U.S. Provisional Patent Application, Serial No. 60/423,009, filed November 1, 2002 and entitled "**METHODS FOR FLEXIBLE FLUID DELIVERY AND HOTSPOT COOLING BY MICROCHANNEL HEAT SINKS**", which is hereby incorporated by reference. In addition, this patent application claims priority under 35 U.S.C. 119(e) of co-pending U.S. Provisional Patent Application, Serial No. 60/442,383, filed January 24, 2003 and entitled "**OPTIMIZED PLATE FIN HEAT EXCHANGER FOR CPU COOLING**", which is hereby incorporated by reference. Additionally, this application claims priority under 35 U.S.C. 119(e) of co-pending U.S. Provisional Patent Application, Serial No. 60/455,729, filed March 17, 2003 and entitled **MICROCHANNEL HEAT EXCHANGER APPARATUS WITH POROUS CONFIGURATION AND METHOD OF MANUFACTURING THEREOF**", which is hereby incorporated by reference.

Field of the Invention

The invention relates to a method of and apparatus for cooling a heat producing device in general, and specifically, to a method of and apparatus for reducing temperature differences and achieving hot spot cooling in a heat source.

Background of the Invention

Since their introduction in the early 1980's, microchannel heat sinks have shown much potential for high heat-flux cooling applications and have been used in the industry. However,

existing microchannels include conventional parallel channel arrangements which are not optimally suited for cooling heat producing devices which have spatially-varying heat loads. Such heat producing devices have areas which produce more heat per unit area than others. These hotter areas are hereby designated as "hot spots" whereas the areas of the heat source which do not produce as much heat are hereby termed, "warm spots". In the simplest case, a hot spot is an area of a heat source, for example a microprocessor, which has a substantially higher heat flux than the other areas of the heat source. In addition, a substantially varying heat flux across the surface of the heat source can induce temperature differences along the heat source surface, thereby forming multiple hot spots.

Figure 1A illustrates a perspective view of a heat source 99 having multiple hot spots therein. As shown in Figure 1A, although the hot spots have a higher heat flux than other areas in the heat source, a peripheral area proximal to the hot spot also has a higher temperature relative to the non-hot spot areas, due to the propagation of heat through the heat source material. Therefore, the area shown within the dashed lines in Figure 1A that is peripheral to the hot spots are higher in temperature than the areas outside of the dashed lines. Therefore, the hot spot area as well as the immediate surrounding area is defined as the hot spot and is called an interface hot spot region.

Alternatively, the heat source 99 does not have any hot spots, as shown in Figure 1B. Figure 1B illustrates a perspective view of a heat source 99 having no hot spots therein along with an aligned graph which represents the temperature variation as a function of distance in the X and Y directions. Although the heat source 99 in Figure 1B does not have any hot spots, the physics of heat propagation in materials dictates that the middle of the heat source 99 will have a higher heat flux than the surrounding areas and edges of the heat source 99. This is shown in the graph in Figure 1B. Prior art heat exchangers only focus on cooling the heat source and thereby do not focus on the aspects of hot spot cooling or overall temperature uniformity.

What is needed is a fluidic cooling loop system with a heat exchanger utilizing various design controls and cooling methods to achieve temperature uniformity in the heat source.

What is also needed is a fluidic cooling loop system with a heat exchanger utilizing various design and control methods to effectively cool hot spots in a heat source.

Summary of the Invention

One aspect of the invention is directed to a method of controlling temperature of a heat source in contact with a heat exchanging surface of a heat exchanger, wherein the heat exchanging surface is substantially aligned along a plane. The method comprises channeling a first temperature fluid to the heat exchanging surface, wherein the first temperature fluid undergoes thermal exchange with the heat source along the heat exchanging surface. The method comprises channeling a second temperature fluid from the heat exchange surface, wherein fluid is channeled to minimize temperature differences along the heat source.

Another aspect of the invention is directed to a heat exchanger for controlling a heat source temperature. The heat exchanger comprises a first layer that is in substantial contact with the heat source. The first layer is configured to perform thermal exchange with fluid flowing in the first layer, wherein the first layer is aligned along a first plane. The heat exchanger comprises a second layer that is coupled to the first layer and channels fluid to and from the first layer. The heat exchanger is configured to minimize temperature differences along the heat source.

Another aspect of the invention is directed to a hermetic closed loop system which controls a temperature of a heat source. The system comprises at least one heat exchanger for controlling the temperature of the heat source. The heat exchanger further comprises an interface layer that is in substantial contact with the heat source and is configured to channel fluid along at least one thermal exchange path, whereby the interface layer is configured along a first plane. The heat exchanger also further comprises a manifold layer which delivers inlet fluid along at least one inlet path and removes outlet fluid along at least one outlet path. The heat exchanger is configured to achieve substantial temperature uniformity in the heat source. The system also comprises at least one pump for circulating fluid throughout the loop. The at least

one pump is coupled to the at least one heat exchanger. The system also comprises at least one heat rejector which is coupled to the at least one pump and the at least one heat exchanger.

In one embodiment, the second layer further comprises a plurality inlet fluid paths which are configured substantially perpendicular to the first plane. The second layer also includes a plurality of outlet paths which are configured substantially perpendicular to the first plane, wherein the inlet and outlet paths are arranged parallel with one another. In another embodiment, the second layer further comprises a plurality inlet fluid paths which are configured substantially perpendicular to the first plane. The second layer also includes a plurality of outlet paths which are configured substantially perpendicular to the first plane, wherein the inlet and outlet paths are arranged in non-parallel relation with one another. In yet another embodiment, the second layer further comprises a first level which has at least one first port that is configured to channel fluid to the first level and a second level having at least one second port. The second level is configured to channel fluid from the first level to the second port, wherein fluid in the first level flows separately from the fluid in the second level.

In the above embodiments, the fluid is in single phase, two phase, or a transition between single and two phase flow. The fluid is channeled along at least one fluid path which is configured to apply a desired fluidic resistance to the fluid The fluid paths include a length dimension and a hydraulic dimension wherein the hydraulic dimension varies with respect to the flow length dimension. The hydraulic dimension is adjustable in response to one or more operating conditions in the heat exchanger. The present invention includes a means for sensing at least one desired characteristic at a predetermined location along the fluid path. The fluid is directed to a first circulation path along a first desired region of the heat exchanging surface. The fluid is also directed to a second circulation path along a second desired region of the heat exchanging surface, wherein the first circulation path flows independently of the second circulation path. One or more selected areas in the heat exchange surface are configured to have a desired thermal conductivity to control the thermal resistance. The heat exchange surface is configured to include a plurality of heat transferring features thereupon, wherein heat is transferred between the fluid and the plurality of heat transferring features. A portion of the heat

exchange surface is roughened to a desired roughness to control at least one of the fluidic and thermal resistances. At least one of the heat transferring features further comprises a pillar, a microchannel and/or a microporous structure. The heat exchange surface includes a desired number of heat transferring features disposed per unit area to control the fluidic resistance to the fluid. In one embodiment, the fluidic resistance is optimized by selecting an appropriate pore size and an appropriate pore volume fraction in a microporous structure. In another embodiment, the fluidic resistance is optimized by selecting an appropriate number of pillars and an appropriate pillar volume fraction in the unit area. In another embodiment, the fluidic resistance is optimized by selecting an appropriate hydraulic diameter for at least one microchannel. The heat transferring feature has a length dimension which is optimized to control the fluidic resistance to the fluid. At least one dimension of at least a portion of the heat transferring feature is optimized to control the fluidic resistance to the fluid. Alternatively, a distance between two or more heat transferring features is optimized to control the fluidic resistance to the fluid. Alternatively, a coating is applied upon at least a portion of at least one heat transferring feature in the plurality to control at least one of the thermal and fluidic resistances. A surface area of at least one heat transferring feature is optimized to control the thermal and fluidic resistances to the fluid. At least one flow impeding element is configured along the fluid path, wherein the at least one flow impeding element controls a resistance. Additionally, a pressure of the fluid is adjusted at a predetermined location along the fluid path to control an instantaneous temperature of the fluid. A flow rate of the fluid is also adjusted at a predetermined location along the flow path to control an instantaneous temperature of the fluid.

Other features and advantages of the present invention will become apparent after reviewing the detailed description of the preferred and alternative embodiments set forth below.

Brief Description of the Drawings

Figure 1A illustrates a perspective view of a heat source having multiple hot spots marked thereon.

Figure 1B illustrates a Temperature-Position graph of a typical heat source having uniform heating.

Figure 2A illustrates a schematic diagram of a closed loop cooling system incorporating a microchannel heat exchanger of the present invention.

Figure 2B illustrates a schematic diagram of a closed loop cooling system incorporating a microchannel heat exchanger with multiple pumps of the present invention.

Figure 3A illustrates a perspective view of the interface layer having several microchannels disposed thereon.

Figure 3B illustrates a perspective view of the interface layer having several different heat transferring features disposed thereon with differing dimensions.

Figure 3C illustrates a perspective view of the interface layer having a varying density of several micro-pillars disposed thereon.

Figure 3D illustrates a perspective view of the interface layer having several micro-pillars and fins disposed thereon.

Figure 4A illustrates an exploded view of one embodiment of the heat exchanger coupled to a heat source in accordance with the present invention.

Figure 4B illustrates a top view of the one embodiment of the heat exchanger with variably moveable fingers coupled to a heat source in accordance with the present invention.

Figure 5 illustrates a cut-away perspective view of another embodiment of the heat exchanger in accordance with the present invention.

Figure 6 illustrates an exploded view of the another embodiment of the heat exchanger in accordance with the present invention.

Figure 7A illustrates a perspective view of another embodiment of the heat exchanger in accordance with the present invention.

Figure 7B illustrates a top view of the one embodiment of the heat exchanger with variably moveable fingers coupled to a heat source in accordance with the present invention.

Figure 8A illustrates a schematic diagram of the thermal resistance to fluid flow circulated to the heat exchanger of the present invention by one pump.

Figure 8B illustrates a schematic diagram of the thermal resistance to fluid flow circulated to the heat exchanger of the present invention by multiple pumps.

Figure 9 illustrates a perspective view of microchannels and micro-pillars having a cut-away feature in accordance with the present invention.

Figure 10 illustrates a Pressure versus Flow Rate diagram of a fluid circulating through a heat exchanger.

Detailed Description of the Present Invention

Generally, a closed-loop fluid system according to the present invention operating in conjunction with a heat exchanger to capture thermal energy generated from a heat source by passing fluid through selective areas of the interface layer in contact with the heat source. In particular, the fluid can be directed in one or two phase flow to specific areas in the interface layer to cool hot spots and/or to reduce temperature differences across the heat source while maintaining an optimal pressure drop within the heat exchanger. In addition, achieving temperature uniformity encompasses minimizing temperature gradients that occur automatically in any heat source. In addition, achieving temperature uniformity in the heat source encompasses minimizing temperature gradients in the absence of hot spots, as in Figure 1B. Therefore, achieving temperature uniformity includes reducing temperatures differences between the hotter areas, warmer areas and cooler areas in the heat source. As discussed below, the heat exchanger as well as the closed loop system of the present invention employ different design concepts and control methods to achieve temperature uniformity throughout the heat source.

It will be apparent to one skilled in the art that although the microchannel heat exchanger of the present invention is described and discussed in relation to cooling hot spot locations in a device, the heat exchanger is alternatively used for heating a cold spot location in a device to achieve temperature uniformity in the heat source. It should also be noted that although the present invention is preferably described as a microchannel heat exchanger, the present invention can be used in other applications and is not limited to the discussion herein.

Figure 2A illustrates a schematic diagram of a hermetically sealed closed loop cooling system 30 which includes the microchannel heat exchanger 100 of the present invention. In addition, Figure 2B illustrates a schematic diagram of an alternative hermetically sealed closed loop cooling system 30' which includes the microchannel heat exchanger 100' with multiple ports 108, 109 coupled to multiple pumps 32' and a diverting valve 33'. The diverting valve 33' and multiple pumps 32' supply more than one fluid stream to the heat exchanger 100'. It should be noted that the system 30, 30' alternatively incorporates additional components not shown in the figures and is not limited to the configuration shown.

As shown in Figure 2A, the fluid ports 108, 109 are coupled to fluid lines 38 which are coupled to a pump 32 and a heat condensor 30. The pump 32 pumps and circulates fluid within the closed loop 30. In one embodiment, a uniform, constant amount of fluid flow enters and exits the heat exchanger 100 via the respective fluid ports 108', 109'. Alternatively, variable amounts of fluid flow enter and exit through the inlet and outlet port(s) 108, 109 of the heat exchanger 100' at a given time. Alternatively, as shown in Figure 2B, two or more pumps 32' provide fluid to several designated inlet ports 108 via one or more valves 33'. It will be apparent that the architectures shown in Figures 2A and 2B are representative only. Any number of pumps and fluid ports can be provided.

As shown in Figures 2A-2B, one or more sensors 130 are coupled to the heat exchanger 100 and/or heat source 99, whereby the sensors 130 provide information of the operating conditions in the heat exchanger 100 to a dynamic sensing and control module 34. The control module 34 is coupled to the pumps 32' and/or heat exchanger 100' and dynamically controls the amount and flow rate of fluid entering and exiting the heat exchanger 100' in response to information received from the one or more sensors 130 regarding changes in heat, hot spot locations, flow rates, fluid temperatures, pressure of the fluid and general operation of the system 30. For instance, the control module 34 initiates operation of both pumps 32' in response to an increase in the amount of heat in a hot spot location. It should be noted that the sensing and control module 34 is applicable to both cooling systems, as shown in Figures 2A-2B.

To better understand the design concepts and methods of the present invention, several heat exchangers are alternatively utilized in the closed loop fluid system 30, 30' and are configureable to incorporate all aspects of the present invention discussed below. The specifics and details of each heat exchanger described below are provided in co-pending patent application Serial No. Cool-01301, filed on October 6, 2003, and entitled "METHOD AND APPARATUS FOR EFFICIENT VERTICAL FLUID DELIVERY FOR COOLING A HEAT PRODUCING DEVICE", which is hereby incorporated by reference. It is apparent to one skilled in the art that the present invention is not limited only to the heat exchangers described in Serial No. Cool-01301 and is applicable to any other appropriate heat exchangers or heat sinks used.

The interface layer 102 (Figures 3A and 4A) is preferably in contact with the heat source and provides heat exchange capabilities to adequately cool the heat source 99. Alternatively, the interface layer 102 is integrally formed within the heat source as one entire component. Alternatively, the interface layer 102 is incorporated into a heat spreader (not shown), whereby the heat spreader is either coupled to or integrally formed within the heat source. The interface layer 102 of the heat exchanger 100 is configured to allow fluid to flow thereupon. The interface layer 102 allows heat transfer from the heat source 99 to the fluid by conduction as well as convection. The interface layer includes any number of similar or different heat transferring features, some of which are described below. It should be apparent to one skilled in the art that the heat transferring features are not limited to the shapes discussed below and alternatively incorporate other appropriate shapes and designs.

Figure 3A illustrates a perspective view of the of the interface layer 102 having several microchannels 110 disposed thereon in accordance with the present invention. The microchannels 110 allow the fluid to undergo thermal exchange along the entire interface layer 102 and/or selected hot spot locations of the interface hot spot region to cool the heat source 99. The microchannel walls 110 extend vertically from the bottom surface of the interface layer and are preferably configured to be parallel, as shown in Figure 3A. Alternatively, the microchannel walls 110 are configured to be non-parallel.

Figure 3B illustrates a perspective view of the of the interface layer 302 having several different heat transferring features disposed along the interface layer in accordance with the present invention. The interface layer 102' includes multiple microchannels 109, wherein two of the microchannels are of the same shape and one microchannel 111 has a portion extending taller than the other portion. In addition, the interface layer 102' includes several pillars 132, 134 of various height dimensions disposed thereon in accordance with the present invention. As shown in Figure 3B, the pillars 134 extend vertically from the bottom surface of the interface layer 302 to a predetermined height, potentially the entire height of the interface layer 102'. The pillars 132 extend vertically an amount less than the pillars 134. The pillars 134 can have any shape including, but not limited to, squared (Figure 3B), diamond (not shown), elliptical (not shown), hexagonal (not shown), circular or any other shape. The interface layer alternatively has a combination of differently shaped pillars disposed thereupon. In addition, Figure 3B illustrates a microporous structure 136 disposed on the bottom surface of the interface layer 102'.

It is preferred that the heat exchanger 100 of the present invention is larger in width than the heat source 99. In the case where the heat exchanger 100 is larger than the heat source 99, an overhang dimension exists. The overhang dimension is the farthest distance between one outer wall of the heat source 99 and the interior fluid channel wall of the heat exchanger 100. In the preferred embodiment, the overhang dimension is within the range of and including 0 to 5 millimeters for single phase and 0 to 15 millimeters for two phase fluid. In addition, the interface layer 102 of the present invention preferably has a thickness dimension within the range of and including 0.3 to 0.7 millimeters for single phase fluid and 0.3 to 1.0 millimeters for two phase fluid.

In the embodiment of the heat exchanger 100 which utilizes a microporous structure 136 disposed upon the interface layer 102, the microporous structure 136 has an average pore size within the range of and including 10 to 200 microns for single phase as well as two phase fluid. In addition, the microporous structure 136 has a porosity within the range and including 50 to 80 percent for single phase as well as two phase fluid. The height of the microporous structure 136

is within the range of and including 0.25 to 2.00 millimeters for single phase as well as two phase fluid.

In the embodiment which utilizes pillars 132, 134 (hereinafter referred to as 132) and/or microchannels 109, 111, 113 (hereinafter referred to as 109) along the interface layer 102, the interface layer 102 of the present invention has a thickness dimension in the range of and including 0.3 to 0.7 millimeters for single phase fluid and 0.3 to 1.0 millimeters for two phase fluid. In addition, the area of at least one pillar 132 is in the range of and including $(10 \text{ micron})^2$ and $(100 \text{ micron})^2$ for single phase as well as two phase fluid. In addition, the area of the separation distance between at least two of the pillars 132 and/or microchannels 109 is in the range of and including 10 to 150 microns for single phase as well as two phase fluid. The width dimension of the microchannels 109 in the range of and including 10 to 100 microns for single phase as well as two phase fluid. The height dimension of the microchannels 109 and/or pillars 132 is within the range of and including 50 to 800 microns for single phase fluid and 50 microns to 2 millimeters for two phase fluid. It is contemplated by one skilled in the art that other dimension are alternatively contemplated.

For instance, as shown in Figure 3D, the interface layer 102" includes several sets of rectangular fins 136 which are radially disposed with respect to one another in their respective set. In addition, the interface layer 302 includes several pillars 134 disposed between the sets of rectangular fins 136. It is apparent that the interface layer 102 can include one type of heat transferring feature or alternatively any combination of different heat transferring features (e.g. microchannels, pillars, micro-porous structures).

The interface layer 102 preferably has a high thermal conductivity which minimizes the temperature differences between the heat source 99 and the fluid flowing along the interface layer 302. The interface layer is preferably made from a material having a high thermal conductivity of 100 W/m-K. The heat transferring features preferably have thermal conductivity characteristics of at least 10 W/m-K. However, it is apparent to one skilled in the art that the interface layer 102 and heat transferring features have a thermal conductivity of more or less than the preferred amount and is not limited thereto. More details regarding the interface layer

as well as the heat transferring features are discussed in co-pending patent application Serial No. Cool-01301, filed on October 6, 2003, and entitled "METHOD AND APPARATUS FOR EFFICIENT VERTICAL FLUID DELIVERY FOR COOLING A HEAT PRODUCING DEVICE", which is hereby incorporated by reference.

The cooling system 30 (Figure 2A) and the heat exchanger 100 of the present invention utilize methods and designs to achieve temperature uniformity and effectively cool hot spots spatially and temporally in the heat source 99'. Figure 4A illustrates an exploded view of one embodiment of the heat exchanger 100 in accordance with the present invention. As shown in Figure 4A, the top surface of the manifold layer 106 is partially cut away to illustrate the channels 116, 122 and fingers 118, 120 within the body of the manifold layer 106. As stated above, the locations in the heat source 99' that produce more heat as well as the region that surrounds that location are hereby designated as interface hot spot regions, whereby the locations in the heat source 99' which produce less heat are hereby designated as warm spot regions. As shown in Figure 4A, the heat source 99' is shown to have hot spot regions at locations A and B. As shown in Figure 4A, the interface layer 102 includes interface hot spot region A, which is positioned above hot spot location A and interface hot spot region B, which is positioned above hot spot location B.

As shown in Figure 4A, fluid initially enters the heat exchanger 100 through one inlet port 108, although more than one inlet port 108 is contemplated. The fluid then flows to an inlet channel 116. Alternatively, the heat exchanger 100 includes more than one inlet channel 116. As shown in Figures 4A and 4B, fluid flowing along the inlet channel 116 from the inlet port 108 initially branches out to finger 118D. In addition, the fluid which continues along the rest of the inlet channel 116 flows to individual fingers 118B and 118C and so on. In the example, fluid is supplied to interface hot spot region A by flowing to the finger 118A, whereby fluid flows down in the Z-direction through finger 118A to the intermediate layer 104. The fluid then flows through an inlet conduit 105A in the interface layer 104 which is positioned below the finger

118A, to the interface layer 102. The fluid preferably travels along the microchannels 110 as shown in Figure 4B and undergoes thermal exchange with the heat source 99'. The heated liquid then travels upward in the Z-direction through the conduit 105B to the outlet finger 120A.

Similarly, fluid flows down in the Z-direction through fingers 118E and 118F to the intermediate layer 104. The fluid then flows through the inlet conduit 105C down in the Z-direction to the interface layer 102. The heated fluid then travels upward in the Z-direction from the interface layer 102 through the outlet conduit 105D to the outlet fingers 120E and 120F. The heat exchanger 100 removes the heated fluid in the manifold layer 106 via the outlet fingers 120, whereby the outlet fingers 120 are in communication with the outlet channel 122. The outlet channel 122 allows fluid to flow out of the heat exchanger 100 through one or more outlet ports 109.

Figure 5 illustrates a broken-perspective view of another embodiment of the heat exchanger 200 in accordance with the present invention. As shown in Figure 5, the heat exchanger 200 is divided into separate regions dependent on the amount of heat produced along the body of the heat source 99". The divided regions are separated by the vertical intermediate layer 204 and/or microchannel wall features 210 in the interface layer 202. Alternatively, the regions in the interface layer 202 are divided by vertical walls which extend between the interface layer and intermediate layer 204, as shown by the dashed lines in Figure 5. However, it is apparent to one skilled in the art that the assembly shown in Figure 5 is not limited to the configuration shown and is for exemplary purposes.

The heat source 99" has a hot spot in location A' and a warm spot, in location B', whereby the hot spot in location A' produces more heat than the warm spot in location B'. It is apparent that the heat source 99" alternatively has more than one hot spot and/or warm spot at any location at any given time. Accordingly, more fluid and/or a higher rate of liquid flow is provided to the interface hot spot region A' in the heat exchanger 200 to adequately cool location A'. It is apparent that although interface hot spot region B' is shown to be larger than interface hot spot region A', interface hot spot regions A' and B', as well as any other interface hot spot regions in the heat exchanger 200, can be any size and/or configuration with respect to

one another. In one embodiment, the heat exchanger 200 is coupled to two or more pumps, as shown in Figure 5, whereby each pump 32' (Figure 2B) provides its own or multiple fluid loops within the heat exchanger 200. Alternatively, each pump 32' (Figure 2B) contributes to one fluid loop which is controllable by the valve 33'. In an alternative embodiment, the heat exchanger 200 is coupled to one pump 32 (Figure 2A).

As shown in Figure 5, the fluid enters the heat exchanger 200 via fluid ports 208A and is directed to interface hot spot region A by flowing along the intermediate layer 204A to the inflow conduits 205A. The fluid then flows down the inflow conduits 205A in the Z-direction into the interface hot spot region A of the interface layer 202. The fluid flows in between the microchannels 210A whereby heat from location A' transfers to the fluid by conduction through the interface layer 202. The heated fluid flows along the interface layer 202 in interface hot spot region A' toward exit port 209A where the fluid exits the heat exchanger 200. It is apparent to one skilled in the art that any number of inlet ports 208 and exit ports 209 are utilized for a particular interface hot spot region or a set of interface hot spot regions.

Similarly, the heat source 99" in Figure 5 has a warm spot in location B' which produces less heat than location A'. Fluid entering through the port 208B is directed to interface hot spot region B' by flowing along the intermediate layer 204B to the inflow conduits 205B. The fluid then flows down the inflow conduits 205B in the Z-direction into interface hot spot region B of the interface layer 202. The fluid flows and is channeled along the microchannels 210, whereby heat generated by the heat source in location B' is transferred to the fluid. The heated fluid flows along the entire interface layer 202B in interface hot spot region B' and upward to exit ports 209B in the Z-direction via the outflow conduits 205B in the intermediate layer 204. The fluid then exits the heat exchanger 200 through the exit ports 209B.

In one embodiment, the heat exchanger 200 is coupled to one pump 32 as shown in the closed loop system 30 (Figure 2A). In another embodiment, the heat exchanger 200 is coupled to more than one pump 32', whereby a set of input ports 208A and output ports 209A are coupled to one pump (pump 1) whereas another set of input ports 208B and output ports 209B are

coupled to another pump 32 (pump 2). Alternatively, the valve 33' (Figure 2B) can direct a different amount of flow to port 208A and 208B.

The heat exchanger 200 is designed in one embodiment to keep a desired fraction of the flow separate such that fluid from one pump does not mix with fluid from another pump. Thus, there is more than one independent fluid loop circulating within the heat exchanger 200. In particular, the heat exchanger 200 in Figure 5 has an independent fluid loop to interface hot spot region A' and another independent fluid loop to interface hot spot region B'. As discussed in more detail below, the independent loops in the heat exchanger 200 are used to achieve temperature uniformity and effectively cool the hot spots in the heat source 99". The independent fluid loops can be used to supply a consistent amount of fluid to one or more interface hot spot region as well as the remaining portion of the interface layer.

Figure 6 illustrates an exploded view of another embodiment of the heat exchanger 300 embodiment in accordance with the present invention. The manifold layer 306 shown in Figure 6 includes three individual levels. In particular, the manifold layer 306 includes a circulation level 304, an inlet level 308 and an outlet level 312. Alternatively, the circulation level 304 is not utilized, whereby the interface layer 302 is coupled directly to the inlet level 308. As shown by the arrows in Figures 6, cooled fluid enters the heat exchanger 300 through the inlet port 315 in the outlet level 312. The cooled fluid travels down the inlet port 315 to the inlet port 314 in the inlet level 308. The fluid then flows into the corridor 320 and flows downward in the Z-direction to the interface layer 302 via the inlet apertures 322 in the circulation level 304. However, the cooled fluid in the inlet corridor 320 does not mix or come into contact with any heated fluid exiting the heat exchanger 300. The fluid entering the interface layer 302 undergoes thermal exchange with the solid material and absorbs the heat produced in the heat source 99. The inlet apertures 322 and outlet apertures 324 are arranged such that the fluid travels the optimal closest distance along the interface layer 302 from each inlet aperture 322 to an adjacent outlet aperture 324. The optimal distance between the inlet and outlet apertures reduces the pressure drop therebetween while effectively cooling the heat source 99. The heated fluid then travels upward in the Z-direction from the interface layer 302 through the inlet level 308 via the

several outlet apertures 324 to the outlet corridor 328 in the outlet level 312. Alternatively, the heated fluid travels upward in the Z-direction from the interface layer 302 directly to the outlet corridor 328 in the outlet level 312. The heated fluid, upon entering the outlet corridor 328 in the outlet level 312 flows to the outlet port 316 and exits the heat exchanger 300. The heated fluid does not mix or come into contact with any cooled fluid entering the manifold layer 306 as it exits the heat exchanger 300. It is apparent that the fluid flow shown by the arrows in Figure 6 is alternatively reversed.

Figure 7A illustrates a perspective view of another embodiment of the heat exchanger in accordance with the present invention. The manifold layer 406 in Figure 7A includes a plurality of interwoven or inter-digitated parallel fluid fingers 411, 412 which allow one phase and/or two-phase fluid to circulate to the interface layer 402 without allowing a substantial pressure drop from occurring within the heat exchanger 400 and the system 30, 30' (Figures 2A-2B). In one embodiment, the inlet fingers 411 are arranged alternately with the outlet fingers 412 in the heat exchanger 400.

In general operation, fluid enters the manifold layer 406 at fluid port 408 and travels through the passage 414 and towards the fluid fingers or passages 411. The fluid enters the opening of the inlet fingers 411 and flows the length of the fingers 411 in the X-direction, as shown by the arrows. In addition, the fluid flows downward in the Z direction to the interface layer 402. As shown in Figure 7A, the fluid in the interface layer 402 traverses along the bottom surface in the X and Y directions and performs thermal exchange with the heat source 99. The heated fluid exits the interface layer 402 by flowing upward in the Z-direction via the outlet fingers 412, whereby the outlet fingers 412 channel the heated fluid to the passage 418 in the manifold layer 406 in the X and Y directions. The fluid then flows along the passage 418 and exits the heat exchanger by flowing out through the port 409.

As stated above, the closed fluid loop 30, 30' (Figures 2A-2B) as well as the heat exchanger 100 can be configured to cool hot spots in the heat source 99 and/or achieve an overall temperature uniformity in the heat source 99. In one embodiment, the present invention effectively cools the hot spots by applying a higher flow rate of fluid and/or colder fluid to an

interface hot spot region. This is initially described above in relation to the heat exchangers 100, 200, 300, 400 shown in Figures 4-7A. For sake of clarity, when referring to all the heat exchangers discussed above, the following discussion will reference heat exchanger 100 generally. However, if specific mention to a particular heat exchanger is needed, the corresponding reference number of that heat exchanger will be denoted.

One method of achieving temperature uniformity in the heat source 99 and effective cooling of hot spots is by controlling the hydraulic and thermal resistances in the heat exchanger 100. Alternatively, another method of reducing temperature differences and achieving effective cooling of hot spots is by configuring the heat exchanger 100 to have variable hydraulic resistance along the manifold layer 106, interface layer 102 and/or intermediate layer 104. Alternatively, another method of reducing temperature variations and achieving temperature uniformity in the heat source 99 is by utilizing multiple pumps or channeling different amounts of flow from one or more pumps to independently cool specific desired areas in the interface layer 102.

Figure 8A illustrates a diagram of the hydraulic or fluidic resistances that the fluid potentially experiences in circulating through the heat exchanger. In the example shown in Figure 4A, the heat exchanger 100 has one interface hot spot region, shown in the left branch in the diagram and one interface warm spot region, shown as the right branch in the diagram. It is apparent to one skilled in the art that the discussion of the resistance diagram in Figure 8A is alternatively applicable to any other heat exchanger and is not limited to the heat exchanger 100 in Figure 4A. Although only one hot spot and warm spot resistance branch is shown in Figure 8A, it is understood that any number of hot spots and cooler spot branches are contemplated.

As shown in Figure 8A, the fluid enters through the inlet 500 and flows through the manifold layer 106 (Figure 4A). The features as well as the configuration of the fluid paths in the manifold layer 106 inherently have hydraulic resistances, denoted as $R_{HOT\_MANIFOLD}$ 502 and $R_{WARM\_MANIFOLD}$ 504. In other words, the fluid experiences resistances $R_{HOT\_MANIFOLD}$ 502 and $R_{WARM\_MANIFOLD}$ 504 in the manifold layer 106 in flowing to the interface layer 102. Similarly, fluid flows through the intermediate layer 104 (Figure 4A), whereby the features and

configuration of the fluid paths in the intermediate layer 106 inherently have some hydraulic resistance, denoted as $R_{HOT\_INTERMEDIATE}$ 506 and $R_{WARM\_INTERMEDIATE}$ 508. Thus, the fluid experiences resistances $R_{HOT\_INTERMEDIATE}$ 506 and $R_{WARM\_INTERMEDIATE}$ 508 in flowing to the interface layer 102. The fluid flows to and along the interface layer 102 (Figure 4A), whereby the features and configuration of the fluid paths in the interface layer 102 inherently have some resistance, denoted as $R_{HOT\_INTERFACE}$ 510 and $R_{WARM\_INTERFACE}$ 512, whereby the fluid experiences resistances $R_{HOT\_INTERFACE}$ 510 and $R_{WARM\_INTERFACE}$ 512 in the interface layer 102. The heated fluid then flows up through the intermediate layer 104 and the manifold layer 106, whereby the heated fluid experiences resistances $R_{HOT\_INTERMEDIATE}$ 514, $R_{WARM\_INTERMEDIATE}$ 516 and $R_{HOT\_MANIFOLD}$ 518, and $R_{WARM\_MANIFOLD}$ 520 in the intermediate layer 104 and manifold layer 106, respectively. The heated fluid then flows out of the heat exchanger 100 through the outlet 522.

Figure 8B illustrates a resistance diagram of the hydraulic resistance that the fluid potentially experiences in circulating through the heat exchanger. Although only one hot spot and warm spot resistance branch is shown in Figure 8B, it is understood that any number of hot spots and cooler spot branches are contemplated. The resistance diagram in Figure 8B is conceptually the same as that shown in Figure 8A, although the two fluid pumps are coupled to the heat exchanger 100. As shown in Figure 8B, Pump 1 circulates fluid to the hot spots, whereby the fluid flowing to and from the interface hot spot region is subjected to the individual resistances in the heat exchanger. Similarly, Pump 2 circulates fluid to the cooler regions (e.g. warm spots), whereby the fluid flowing to and from the cooler regions is subjected to the individual resistances in the heat exchanger.

With regard to Figure 8A, the heat exchanger 100 of the present invention can be configured such that the hydraulic and thermal resistances are independently or collectively controlled to reduce temperature differences and achieve temperature uniformity in the heat source 99 and effective hot spot cooling. For example, more than one pump is employed in the system 30' (Figure 2B), whereby more and/or colder fluid is channeled to the hot spot region and less and/or warmer fluid is channeled to the warm spot regions. Therefore, the heat exchanger 100 is configured to have a lower hydraulic resistance along the hot spot branch, $R_{HOT\_MANIFOLD}$

502, $R_{HOT\_INTERMEDIATE}$ 506, $R_{HOT\_INTERFACE}$ 510, whereas the heat exchanger is configured to have a higher resistance in the warm spot branch, $R_{WARM\_MANIFOLD}$ 504, $R_{WARM\_INTERMEDIATE}$ 516 and $R_{WARM\_INTERFACE}$ 512. The thermal resistance is also controllable to allow the fluid channeled to the hot spot to undergo better thermal exchange and heat absorption than fluid channeled to the cooler region, as discussed in more detail below.

This is shown specifically in the heat exchanger 200 in Figure 5 as well as the resistance diagram in Figure 8B, in which one pump (Pump 1) circulates fluid to interface region A and another pump (Pump 2) circulates fluid to interface region B. Although the fluid from the multiple pumps can be mixed at any point in the heat exchanger 200, the two fluid lines are alternatively kept separate from one another within the heat exchanger 200. Therefore, multiple independent cooling loops are established in the heat exchanger 200, whereby the characteristics of the heat source cooling and fluid flow are controllable to achieve effective cooling and reduce temperature variations.

The methods and designs in controlling or varying the hydraulic and thermal resistances are discussed in more detail below. It should be noted that the hydraulic resistance is controllable in any of the layers or levels in the heat exchanger and the thermal resistance is controllable in the interface layer. In addition, it is apparent to one skilled in the art that any combination of the designs and methods are alternatively incorporated in the system and heat exchanger to reduce temperature differences and achieve temperature uniformity in the heat source 99.

The individual features and designs in the manifold layer 106 and intermediate layer 104 can be individually or collectively varied to control the hydraulic resistance in the heat exchanger 100. In the manifold layer 106, the geometries and cross-sectional dimensions of the fingers 118, 120 and channels 116, 122 are tailored to provide a specific hydraulic resistance. In the intermediate layer 104, the geometries and cross sectional dimensions of the conduits 105 are tailored to provide a specific desired hydraulic resistance. For example, a particular finger 118, 120 in the manifold layer 106 preferably has a larger cross sectional dimension along the location above the interface hot spot region, whereas the fingers 118, 120 above the warm

interface region has a smaller cross sectional dimension. This cross sectional area variation enhances cooling capacity to hot spots. As shown in Figure 4B, the width or hydraulic dimension of the channels 116, 122 are variable along the length of the manifold layer 106. In one embodiment, the larger hydraulic dimensions of the channels 116, 122 are positioned above the interface hot spot regions and the narrower hydraulic dimensions of the channels 116, 122 are positioned above the cooler regions. Therefore, more flow is allowed to pass over the hot spot than over the warm spot at a given time. In one embodiment, as shown in Figure 4B and 7B, the channels and/or fingers have fixed or varying hydraulic dimensions which are permanently designed into the manifold layer 106. Alternatively, the variable hydraulic channels and/or fingers are dynamically adjustable as discussed in more detail below.

The vertical dimensions of the fingers 118, 120 and/or channels 116, 122 are alternatively variable to accommodate hot spot cooling and temperature uniformity in the heat source 99. Channels 116, 122 and/or fingers 118, 120 in the manifold layer 106 which have longer side walls allow more fluid to directly travel vertically to the interface layer 102. In contrast, fingers in the manifold layer 106 that do not have vertical walls which extend down to the interface layer 102 allow more fluid to travel horizontally across the interface layer instead of directly striking the interface layer 102. For example, a section of a finger 118, 120 above an interface hot spot region has longer side walls than other sections of that particular finger 118, 120. Thus, the longer side walls allow fluid to be more directly applied to a concentrated area in the interface hot spot region, whereas the shorter side walls allow the fluid to be applied over a greater area in the interface layer. It should be noted that the above discussion of the varying dimensions of the fingers 118, 120 and channels 116, 122 also apply to the other embodiments discussed and is not limited to the embodiment discussed above.

In addition, the heat exchanger 100 is alternatively designed to control the hydraulic resistance $R_{INTERFACE}$ in the interface layer 102 to accommodate temperature uniformity and hot spot cooling in the heat source 99. Preferably, the heat transferring features in the interface layer 102 which are located above or near the interface hot spot regions will have less hydraulic resistance than heat transferring features in other areas in the interface layer 102. Therefore,

more fluid is allowed to pass over the interface hot spot region than over the other areas of the interface layer at a given time, because less hydraulic resistance is present at the interface hot spot region.

The hydraulic resistance in a fluid pathway in the interface layer 102 is controlled by optimizing the hydraulic dimensions of the heat transferring features. For instance, the hydraulic diameter of microchannels 110 are configureable to control the fluid flow rate along the length of the microchannels 110. Thus, one or more microchannels in the interface layer 102 has a larger diameter over the interface hot spot regions than the remaining portions of the microchannels 110. Therefore, the larger diameter microchannels 110 will allow more fluid to pass over the interface hot spot regions than the smaller diameter microchannels 100 having more resistance. Alternatively, the pillars 134 are positioned apart from each other to control the amount of hydraulic resistance upon the fluid flowing along the interface layer 102. Thus, each of the pillars 134 at an interface hot spot region can be positioned farther apart than pillars above a warm spot region, such that more fluid is able to flow over the interface hot spot region than the warm spot region at a given time. It is apparent that the dimension of the hydraulic diameters should be optimized in light of the amount of pressure drop created in the interface layer 102 and the amount of surface area for conduction provided by the heat transferring features.

The hydraulic resistance in a fluid pathway along the interface layer 102 can alternatively be altered by optimizing the length of the fluid pathway. It is well known that the amount of hydraulic resistance increases as the length of the fluid path increases. Therefore, the length of the fluid path can be optimized to minimize the hydraulic resistance along the interface layer 102 while maintaining the pressure characteristics of the fluid. In one example, microchannels 110 located at an interface hot spot region have a smaller channel length in comparison to microchannels 110 at a warm hot spot region. Therefore, fluid traveling over the interface hot spot region flows a shorter distance along the microchannel 110 fluid paths and experiences less hydraulic resistance before exiting the interface layer, whereas the longer microchannels 110 force the fluid to travel a longer distance and cause the fluid to gradually heat up while flowing

along the interface layer. It should be noted that although the length of the fluid path is optimized for single phase flow, the length of the fluid path of the microchannel 110 is alternatively made longer to induce two phase flow, as discussed in more detail below.

In another embodiment, the heat exchanger 100 can be configured to control the thermal conductivity characteristics in the interface layer 102 to accommodate temperature uniformity and hot spot cooling in the heat source. In particular, the heat transferring features discussed above are configured to control the ability to transfer to heat from the heat transferring features to the fluid. Thus, the heat transferring features as well as the interface layer 102 itself can be configured to have one or more locations in the interface layer 102 which have a higher thermal conductivity compared to other locations in the interface layer 102.

One application of controlling the thermal conductivity in the interface layer 102 and/or heat transferring features 110 is to form the interface layer 102 and/or heat transferring features 110 out of appropriate materials which have corresponding thermal conductivity values. For instance, a heat source 99 not having any hot spots will generate a higher heat flux in the center, as shown in Figure 1B. To achieve temperature uniformity in the heat source 99 shown in Figure 1B, the interface layer 102 and/or heat transferring features are formed to provide a higher thermal conductivity in the center of the interface layer 102. In addition, the thermal conductivity properties of the interface layer 102 and/or heat transferring features 110 gradually decreases away from the center, such that the entire heat source 99 is cooled to a substantially uniform temperature.

In addition, the thermal resistances in the heat exchangers 100 are alternatively controlled by selectively adjusting the surface to volume ratio of the heat transferring features in the interface layer 102. By increasing the surface to volume ratio of the features or interface layer 102 itself, the thermal resistance of the features and/or interface layer 102 is reduced. One example of increasing the surface to volume ratio within the interface layer 102 includes configuring the interface layer 102 to have more a greater density of heat transferring features per unit area. For example, as shown in Figure 3B, the microchannels 110 and 111 are positioned closely to one another whereas microchannels 113 are located a further distance away

from microchannels 110 and 111. The microchannels 110 and 111 will provide less thermal resistance to the fluid than the spaced apart microchannels 113 due to the greater surface to volume ratio of heat transferring features in the interface layer 102'. In an application in which a microporous structure 136 (Figure 3B) is disposed on the interface layer 102', the thermal resistance of the microporous structure is reduced by utilizing smaller pore sizes.

In another example shown in Figure 3C, the heat source 99 has a hot spot in each corner. Thus, the interface layer 102" correspondingly includes an interface hot spot region in each corner as shown in Figure 3C. The interface layer 102" can be configured to achieve temperature uniformity in the heat source 99 by reducing the thermal resistance along the outer corners of the interface layer 102". The interface layer 102" thereby has a greater number of pillars 134 positioned near the outer edges of the bottom surface 101, whereby a smaller density of pillars 134 are positioned near the center, as shown in Figure 3C. The greater density of pillars 134 thereby provides a greater surface to volume ratio and a lower thermal resistance in the outer corners of the interface layer 102". It should be noted that the design shown in Figure 3C is only one example and is not limited to the design shown therein. It should be also noted that the dimensions and volume of pillars 134 are optimized such that the fluidic resistance along the interface does not become larger than the thermal resistance.

Another example of increasing the surface to volume ratio within the interface layer 102 is to design the heat transferring features 110 at or near the interface hot spot region to have a vertical dimension that is larger than the vertical dimension of other features in the remaining areas of the interface layer 102'. As shown in Figure 3B, the heat source 99 has a large hot spot along the front half of the body. Accordingly, to achieve effective cooling of the heat source 99, the microchannel 111 and pillars 134 have a greater vertical height near the front half of the interface layer 102', whereas the microchannel 111 and pillars 132 have a smaller vertical height near the rear half of the interface layer 102'.

In two phase flow scenarios, the surface to volume ratio of the heat transferring features are alternatively increased by modifying the shape of the feature to have a greater surface area which the fluid is in contact with. For example, as shown in Figure 9, the microchannels 600

include a longitudinal slot 604 extending into the side of the walls 600. In addition, the pillars 602 include a notch 606 cut out from the body of the pillar 602. The slots 602 in the microchannels 600 provide additional surface area for the fluid to come into contact with. Similarly, the notches 606 in the pillars 602 provide additional surface area for the fluid to come into contact with. The additional surface area provides more space for heat to transfer to the fluid, thereby reducing the thermal resistance in the interface layer. The additional surface area from the slots 604 and notches 606 reduces superheating and promote stable boiling of the fluid in the vicinity of the hot spots in two phase flow. It is apparent that the heat transferring features alternatively have any other configuration to provide an increased surface area to the fluid and the increased surface features shown on the microchannels 600 and pillars 602 in Figure 9 are exemplary.

In two phase flow scenarios, the heat transferring features are additionally configureable such that the surfaces of the heat transferring features are roughened a certain degree at locations where more heat transfer is desired. A roughened surface creates pockets which bubbles from the liquid form within, whereby surface tension along the surface holds the bubbles to the roughened surface. For instance, by changing the roughness of a microchannel walls 110, the surface tension along the microchannel walls 110 is changed, thereby increasing or decreasing the amount of vapor pressure needed to initiate boiling of the liquid. A surface which is substantially rough will require less vapor pressure to initiate boiling, whereas a substantially smooth surface will require more vapor pressure to initiate boiling. In a two phase flow scenario, boiling is desired at interface hot spot regions to achieve effective cooling of the hot spot, as discussed in more detail below. Therefore, the heat transferring features 110 as well as the interface layer 102 can have a roughened surface achieve effective cooling of the hot spot.

The desired surface or surfaces in the interface layer 102 are roughened using conventional surface altering methods. Alternatively, the desired surface or surfaces in the interface layer 102 are roughened by applying a coating to the desired surface. The surface coating applied to the interface layer 102 and/or heat transferring elements 110 modifies the surface tension of the surface. In addition, the surface coating is alternatively applied to modify

the contact angle at which the two phase fluid comes into contact with the surface. The surface coating is preferably the same material applied to alter the thermal conductivity of the interface layer 102 and/or heat transferring features, whereby the thermal conductivity of the coating is at least 10 W/m-K. Alternatively, the surface coating is made of a material different than the material of the interface layer 102.

In addition to controlling the cooling ability of the heat exchanger 100 by altering the thermal and hydraulic resistances, the heat exchanger 100 also achieves temperature uniformity and hot spot cooling in the heat source 99 by exploiting the temperature-dependent viscosity characteristics of the fluid. As known in the art, the viscosity of most fluids decreases with increasing temperature, whereby the fluid becomes less resistive to flow as the fluid temperatures increases. Therefore, hotter areas in the interface layer 102 will naturally draw more fluid thereto than cooler areas due to this reduced hydraulic resistance and viscosity.

In one embodiment, the heat exchanger 100 of the present invention utilizes this property of the fluid in its design. In particular, the heat exchanger initially directs the fluid to the interface hot spot regions, wherein heat transfer from the hot spots will naturally cause the fluid to increase in temperature. As the temperature of the fluid increases, the fluid itself will become less viscous. For example, the heat exchanger 100 is configureable to initially channel fluid to hotter areas in the interface layer 102 to increase the fluid temperature. The heated, less viscous fluid is then channeled at a faster flow rate to the remaining areas of the interface layer 102. Although the fluid is heated to reduce its viscosity, the heating of the fluid can cause the fluid to boil and accelerate as vapor, thereby causing a substantial increase in the pressure drop along the interface layer 102. In one embodiment, the heat exchanger 100 compensates for the potential pressure drop by constricting the flow and preventing the fluid from accelerating. This is performed using a variety of methods, such as designing the fluid paths to have very narrow pores, channels and/or spaces between the heat transferring features or utilizing multiple pumps, as discussed above. In another embodiment, as discussed below, the fluid is purposely allowed to undergo boiling to further cool desired areas in the interface layer 102.

In another embodiment, the heat exchanger 100 of the present invention includes an internal valving mechanism to achieve temperature uniformity and perform effective cooling of hot spots in the heat source 99. In particular, the internal valving mechanism in the heat exchanger 100 controls the fluid flow to selected regions in the interface layer 102. The internal valving mechanism in the heat exchanger 100 dynamically controls the hydraulic and thermal resistance along the fluid path to achieve desired cooling effects in the system 30, 30' (Figure 2A-2B). It is apparent to one skilled in the art that the internal valving mechanism of the heat exchanger 100 allows the system 30, 30' to control the fluid flow rate as well as the amount of flow in the heat exchanger 100. Further, the internal valving mechanism is alternatively utilized to control the phase characteristics as well as any pressure-dependent or viscosity dependent characteristics of the fluid in the heat exchanger 100.

Figures 4B and 7B illustrate alternative embodiments of the manifold layer 106', 406' having multiple internal valves configured within. As shown in Figure 4B, the manifold layer 106' includes an expandable valve 124' along the channel wall 116' near the inlet port 108' as well as another expandable valve 126' which extends around the corner along the inlet channel 116'. In addition, the manifold layer 106' includes an expandable valve 128' in the outlet finger. The valves 124' and 128' are shown in Figure 4B to be expanded, whereas the valve 126' is shown to be contracted. The fluid experiences higher hydraulic resistance at valves 124' and 128' due to the reduced fluid path dimension which the fluid can flow through. In addition to constricting the flow at the valve 124' location, the expanded valve 124' also controls the flow rate as well as the amount of flow which is channeled to the remaining portion of the inlet channel 116'. For instance, the amount of fluid will increase at aperture 119' as the valve 124' is contracted, because the fluid path at the valve 124' is increased in dimension. The expanded valve 128' in the outlet finger 120' also controls the flow rate as well as the amount of flow which is channeled to the remaining portion of the outlet finger 120'. The valve 126', as shown in Figure 4B, provides smaller hydraulic resistance to the fluid than valve 124', although the valve 126' is also expandable to increase the hydraulic resistance to the fluid.

As shown in Figure 7B, the manifold layer 406' includes expandable valves 424' and 428' coupled to the inside walls of inlet fingers 411'. In addition, the manifold layer 406' includes an expandable valve 426' coupled to one side in the outlet finger 412'. Although the some of the valves in Figure 4B and 7B are shown to be entirely expanded or contracted, portions of the valve 424' alternatively expand and/or contract independently of one another. For example, as shown in figure 7B, one side of the valve 424' is expanded whereas the other side of the valve 424' is contracted. In contrast, the entire valve 426' in figure 7B is substantially expanded, whereas the entire valve 428' is contracted. Although not shown, the expandable valves are alternatively disposed along any channel or fluid path in the manifold layer. Although not shown, one or more expandable valve are alternatively configured within the apertures 322, 324 in the heat exchanger 300 in Figure 6. Alternatively, the expandable valves are configured in the conduits 105 in the intermediate layer 104. Alternatively, the expandable valves are configured along the interface layer 102. In addition, the valve can be placed uniformly along the wall surface, as shown with valve 428' in Figure 7B. Alternatively, the valve can be disposed on the wall surface non-uniformly, such as several bumps or other shaped protrusions which are individually expandable and contractible. The individual protrusion-like valves are alternatively useable to selectively increase the surface to volume ratio in the interface layer 102. It should be noted that fixed or variable valves are also applicable to the embodiment shown in Figure 6 although not shown.

In one embodiment, the expandable valve is a shape memory alloy or a differential thermal expansion element. In another embodiment, the expandable valve is a conventional or MEMS type valve. The expandable valve is alternatively made of a temperature driven bi-material which senses the temperature difference and automatically contracts or expands in response to the temperature difference. The expandable valve is alternatively made of a thermo-pneumatic material. The valve alternatively has a bladder configuration which contains expandable organic material having a high expansion coefficient. In another embodiment, the expandable valve is a capacitive valve which actively deflects between a contracted and expanded state to deliver or restrict the amount of fluid to a desired area.

As stated above, the cooling system 30, 30' (Figures 2A-2B) utilizes sensors 130 in the heat exchanger 100 to dynamically control the one or more pumps 32' (Figure 2B) and/or valves inside or outside of the heat exchanger 100. As stated above, the heat source 99 alternatively has characteristics in which the locations of one or more of the hot spots change due to different tasks required to be performed by the heat source 99. In addition, the heat source 99 alternatively has characteristics in which the heat flux of the one or more of the hot spots change over time due to different tasks required to be performed by the heat source 99. The sensors 130 provide information to the control module 34 including, but not limited to, the flow rate of fluid flowing in the interface hot spot region, temperature of the interface layer 102 in the interface hot spot region and/or heat source 99 and temperature of the fluid.

To achieve temperature uniformity and effectively cool the heat source 99 in light of these spatial and temporal heat flux changes, the system 30, 30' includes a sensing and control module 34, 34' (Figures 2A-2B) which dynamically changes the amount of flow and/or flow rate of fluid entering the heat exchanger 100 in response to information provided by the sensors 130.

Figures 2A and 2B illustrate the heat exchanger 100 having multiple sensors 130, 130' placed within to sense the conditions of the heat source 99 as well as provide other information to the control module 34, 34'. In one embodiment, the one or mores sensors 130 are placed in the interface layer 102 and/or alternatively the heat source 99 at any desired location. As shown in Figures 2A-2B, the sensors 130, 130' and control module 34, 34' are also coupled to the one or more pumps 32' (Figure 2A-2B), whereby the information provided by the sensors 130 to the control module 34 actively controls the pump 32. The plurality of sensors 130' are coupled to the control module 34', whereby the control module 34' is preferably placed upstream from heat exchanger 100, as shown in Figures 2A-2B. Alternatively, the control module 34 is placed at any other location in the closed loop system 30. For instance, one pump 3' which is operating at a lower power will increase its output upon receiving information that a particular region in the interface layer 102 is increasing in temperature, thereby causing more fluid to be delivered to that particular region. In the case of multiple pumps 32' (Figure 2B) coupled to one or more valves within or outside the heat exchanger 100, the sensors 130' and control module 38'

alternatively control the flow of fluid to the desired interface hot spot regions via the one or more valves. For example, the expandable valve 426' shown in Figure 7B can be configured to expand or contract in response to information provided by the sensors 130.

In addition to the above designs and methods employed within the system of the present invention, the heat exchanger 100 alternatively employs pressure-dependent boiling point conditions to achieve temperature uniformity and effective cooling of the hot spots in the heat source 99. Depending on the flow characteristics of the fluid in the heat exchanger 100, it is advantageous to subject the interface hot spot regions to fluid in the single, liquid phase or under two-phase, boiling conditions.

For single phase fluids, such as liquids, it is preferred that colder fluid is delivered at a high flow rate to the interface hot spot regions using the designs described above. For two phase fluids, such as a mixture of vapor and liquid, one method of effective cooling of the hot spots is to cause the fluid to boil at the hot spot to effectively cool the hot spot. It is well known that the temperature and boiling point of a two-phase fluid is directly proportional to the pressure of the fluid. In particular, as the amount of pressure in the fluid increases, the temperature and boiling point of the fluid increases. In contrast, as the amount of pressure decreases in the fluid, the temperature and boiling point of the fluid decreases. The heat exchanger 100 utilizes this pressure/temperature phenomenon of the fluid under single or two phase flow to effectively cool the hot spots and achieve temperature uniformity in the heat source 99.

For single phase flow, the heat exchanger 100 is configured to channel fluid that is a relatively low pressure and temperature to one or more desired interface hot spot regions, whereas the heat exchanger 100 simultaneously channels fluid to other parts of the interface layer 102 which is at a relatively higher pressure and temperature. The lower temperature fluid subjected to the hot spots will effectively cool the hot spots to a desired temperature while the higher temperature fluid will cool the warm or cold spots to the same desired temperature. In effect, the single phase flow achieves temperature uniformity in the heat source 99 by directing fluid at the adequate temperature to the desired locations in the interface layer 102 to effectively cool the locations to a desired temperature.

For two phase flow, the heat exchanger 100 of the present invention is configured to channel fluid using the same pressure-temperature phenomenon discussed above. In particular, the heat exchanger 100 of the present invention supplies lower pressure fluid to the interface hot spot regions to purposely create a pressure drop at the interface hot spot regions. It is well known that boiling of a two phase fluid causes a significant pressure drop due to a substantial increase in acceleration of the two phase fluid. As stated above regarding the pressure-temperature relationship, a significant drop in fluid pressure will naturally cause the temperature to significantly drop to a temperature corresponding with the reduced pressure. Accordingly, the heat exchanger 100 is configureable to channel two phase fluid already at a relatively lower pressure to the interface hot spot regions. In addition, the heat exchanger 100 is configureable to channel fluid at a relatively higher pressure to cooler areas of the interface layer 102. The lower pressure fluid, upon coming into contact with the interface hot spot region, will significantly heat up and begin to boil at a much lower boiling point, thereby generating a pressure drop. As a result of the decrease in pressure, the temperature of the boiling two phase fluid effectively decreases. As a result, the two phase fluid becomes cooler and is able to more effectively cool the hot spot. It is apparent that the same theory applies in the reversing two phase fluid into single phase fluid to achieve temperature uniformity in the heat source 99.

In another embodiment, the heat exchanger 100 of the present invention achieves temperature uniformity along the entire heat source 99 using multiple operating points of single and two-phase fluids. Figure 10 illustrates a graph of pressure drop versus flow rate of fluid in a typical heat exchanger coupled to a microprocessor chip. As shown in Figure 10, the pressure of the fluid flowing along the interface layer 102 in the liquid region linearly increases with the flow rate. However, as the fluid flow rate decreases, the fluid undergoes enters the boiling regime and undergoes two-phase flow. As the fluid flow rate decreases in the boiling regime, the pressure of the fluid non-linearly increases. In addition, at significantly lower flow rates, the pressure of the fluid substantially increases in which the fluid at the significantly lower flow rates begins to dry up.

As stated above, the pressure of the fluid is directly proportional to the temperature of the fluid. In addition, as shown in Figure 10, the pressure of the fluid has a relationship with the flow rate of the fluid. Thus, the temperature as well as the boiling point of the fluid is controllable by controlling the flow rate and/or pressure of the fluid.

The heat exchanger 100 of the present invention utilizes multiple fluid conditions to effectively achieve temperature uniformity in the heat source 99. The heat exchanger 100 is configureable to control the cooling effect of the fluid in each desired area by manipulating the fluid flow rate and/or the pressure of the fluid in the desired area using one pump 32 (Figure 2A). Alternatively, heat exchanger 100 controls the cooling effect of the fluid in each desired area by manipulating the fluid flow rate and/or the pressure of the fluid in the desired area using multiple pumps 32' (Figure 2B).

In particular, the heat exchanger 100 controls the pressure and/or flow rate of fluid in desired fluid paths to produce different desired effects in specific areas of the interface layer 102. In relation to the graph in Figure 10, fluid which flows undergoing thermal exchange at a flow rate below 30 ml/min will undergo two phase flow. In contrast, the fluid flowing at a flow rate above 40 ml/min will be in the liquid regime and remain in single phase. For example, referring back to Figure 5, the heat source 99 has a hot spot region in Location A and a warm spot region in Location B. In this particular example, the heat exchanger 200 does not allow fluid flowing in the interface hot spot region A to come into contact with fluid flowing in the interface hot spot region B. It is apparent to one skilled in the art that the several fluid paths do not need to be separated. Thus, the flow rate of all the fluid is increased and decreased as desired along the entire fluid path within the heat exchanger 100 by altering the hydraulic and thermal resistances in the heat exchanger 100. It should be noted that the discussion herein related to the heat exchanger 200 is for exemplary purposes and is applicable to any of the heat exchangers discussed as well as any number of hot/warm spots in the heat source 99.

As stated above regarding the pressure-temperature relationship, flow undergoing transition between single and two phase flow will decrease in temperature due to the pressure drop generated from the boiling of the fluid. Accordingly, the heat exchanger 200 channels two

phase fluid to the interface hot spot region A and simultaneously channels single phase fluid to interface warm spot region B to bring the entire heat source 99 to a uniform temperature. The heat exchanger 200 in the present invention is able to accomplish this effect by channeling pressurized fluid at a lower flow rate to interface hot spot region A and channeling fluid at the same pressure to interface warm spot region B at a higher flow rate. In the present example, the heat exchanger 200 channels fluid at 1 psi to the interface hot spot region A at a flow rate of 20 ml/min, whereby the fluid has two phase characteristics. Simultaneously, the heat exchanger 200 channels fluid at 1 psi to interface warm spot region B at a flow rate of 40 ml/min, whereby the fluid has single phase characteristics.

The flow rate of the fluid is controlled by using any of the designs and methods discussed above relating to hydraulic resistance. For instance, the heat transferring features in the interface layer 102 support different flow rates by controlling the fluid flow along the interface layer 102. Additionally, configurations in the fingers, channels and/or apertures are alternatively optimized to control the flow rate of the fluid. Alternatively, the heat exchanger 100 is configured to provide more effective heat transfer to the fluid using any of the methods and designs discussed above regarding controlling the thermal resistance upon the fluid flow. Alternatively, the heat exchanger 100 is coupled to more than one pump, whereby the multiple pumps 32' circulate independent fluid loops which have different operating conditions in the heat exchanger 100. It is also noted that the same phenomena applies to the heat exchanger 100 keeping the flow rate of the fluid constant whereas the pressure of the desired fluid paths are altered or controlled to achieve the same effects.

The present invention has been described in terms of specific embodiments incorporating details to facilitate the understanding of the principles of construction and operation of the invention. Such reference herein to specific embodiments and details thereof is not intended to limit the scope of the claims appended hereto. It will be apparent to those skilled in the art that modifications may be made in the embodiment chosen for illustration without departing from the spirit and scope of the invention.